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White Oak, Silver Spring, Maryland

NAVAL ORDNANCE LABORATORY MEMORANDUM 10248

10 May 1949

From: TM-4V (W.C. Brueggeman)
(J.T. Hart)
TO: NOL File
Via: TM-O (L.W. Ball)
SUBJ: Multigee Vibrator--A Proposed New Vibration Testing
Machine for Producing High Acceleration
Preliminary Design of. (Project NOL-26Re2b-281-10)

Abstract: The design of a vibration testing machine employing two contra-rotating driver units containing mercury as the variable eccentric mass has been completed. The unbalance and resultant centrifugal force of the units is controlled by forcing mercury from one to the other of two connected opposed chambers in each unit by means of compressed air under controlled pressure; this pressure also maintains steady-state balance between the mercury in the two chambers. Theoretically the acceleration of the table will remain nearly constant for a given pressure and load regardless of the speed. The computed performance of the machine at a 20 lb load is .073 amplitude for speeds up to about 1600 cps and about 20g acceleration for speeds between 1900 and 3600 cpm.

Foreword: This is a Divisional NOLM describing a machine designed by the authors. One such machine is under construction at the date of this report. The details of construction must be regarded as tentative and the computed performance subject to verification until the machine has been tested in operation.

Refs: (a) TSS 5383
(b) NOLM 8778 - Proposed Machine for Safety Tests of Ammunition.
(c) TSS 4852
(d) Den Hartog, Mechanical Vibrations

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NOLM 10248

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- (e) NOLM 8860 - Standard (Laboratory) Transportation Vibration Tests and Special Design Vibration Tests as Used for NOL Ordnance.

Encls: (A) Plates 1 through 8.

1. This machine, which has been designed in compliance with Reference (a), is intended to meet the need for high-acceleration vibration testing facilities pointed out in Reference (b). Vibration at a peak acceleration of 10 to 20g, to determine handling safety, is indicated as one of a series of mechanical tests which the author of NOLM 8778 believes necessary for all small ammunition devices associated with explosives. Tests on representative fuzes of existing designs in a range of accelerations above the 2g involved in the Standard Transportation Vibration Test have confirmed the possibility of indicating unsafety by such accelerated, over-simulating tests (Reference (c)).

2. If the predicted performance is realized, the subject machine will also be suitable for general use in any vibration test requiring linear motion of the table perpendicular to its plane, and it is anticipated that this type of machine will find application for testing many other types of mechanisms. This machine employs a construction and principle of operation which is believed to be new in the field of vibrator design and apparently offers several important advantages in range of performance, ease of control and simplicity of construction. Thus, in addition to filling a specific need for a high-acceleration vibrator, this machine is to be considered as a prototype model for experimentation with and further development of vibrators of this new type.

Performance Requirements

3. It was desired to provide a single amplitude of vibration perpendicular to the vibrator table of at least 0.32 throughout the range 700-1500 cpm for a 20 lb load, and sufficient amplitude to produce 20g peak acceleration throughout the range 1500-3600 cpm. The frequency range of 700-3000 cpm covers that required in the Standard Laboratory Transportation Vibration Test, Reference (e). Since a machine capable of higher frequency would be useful, the design provides for operation (with slight modification) up to 5000 cpm at somewhat lower acceleration. The performance of commercial anti-friction bearings limits the frequency at full capacity to about 3600 cpm.

Principle of Operation

4. It was decided that the vibrator should be of the reaction type as this type is relatively simple (compared to the cam-driven) in construction and requires no special foundation. The three of this type (NOL designed) presently installed in the vibration laboratory have given satisfactory performance. These have a single, adjustable-while-running driver which is attached to a table mounted on four compression springs. It was felt that due to the relatively great amplitude of vibration required of the new machine a pair of contra-rotating drivers should be used, thus making it possible to balance out the lateral component of the inertia force. This should result in a well-defined straight-line motion. Elimination of the lateral component associated with the single driver should result in greater stability of the table, especially for loads having a high center of gravity.

5. In considering the design of the rotating eccentric-mass driving units, it appeared that two units of the type presently employed on NOL vibrators having the required capacity would be excessive in size and weight and would require a rather cumbersome mechanical linkage for adjustment of the eccentrics. After considering various possibilities, the scheme shown in Plates 1 to 4 was finally adopted. Each of the contra-rotating drivers contains two chambers which are identical in dimensions, diametrically opposed and symmetrically located with respect to the axis of rotation. Plates 1, 2 and 3 show the position of these chambers (labeled A and B in Plate 3). The two chambers in each driver are connected by a circular passageway; the chambers contain mercury. If the drivers are rotated at a speed which produces radial acceleration exceeding $1g$, the mercury will remain in the outermost portion of the chambers and will divide equally between the two chambers if both are under atmospheric pressure, (Plate 3, top). The centrifugal forces developed in both chambers will balance giving zero resultant centrifugal force. If the pressure of the air in chamber A is increased above atmospheric, some of the mercury will be forced from Chamber A to Chamber B, (plate 3) assuming that Chamber B is vented to the atmosphere. This redistribution of the mass will cause a resultant centrifugal force approximately equal to the longitudinal cross-sectional area of the chamber times the air pressure in Chamber A for each driver. Each driver of the pair is connected to the same source of controlled pressure. When geared together to rotate in opposite directions, the resultant centrifugal force of the two equal drivers is zero in the plane containing their axes and twice the component of each in a direction perpendicular to this plane.

6. Details of construction of the drivers are shown in Plate 1. A view of the entire machine is shown in Plate 4. As shown in Plate 1, the drivers are constructed of welded and machined parts and are simple and rugged. The compressed air is brought in through a rotating pressure joint employing a flanged Oilite bearing. Failure or release of the air pressure will immediately produce a balanced condition with zero force output. If the pressure is increased above the amount needed to empty chamber A the air will pass through the communicating tube and bubble out to the atmosphere through the mercury in B. When the machine is at rest some of the mercury will run into the central cavities, which are of sufficient volume to accommodate all of the mercury without danger of overflow through the vent hole or air inlet. This mercury will be forced back into the chambers A and B as soon as rotation is started.

Advantages

7. It is obvious that the resultant centrifugal force of the two drivers is a sine function of the angular position of the drivers, is directly proportional to the air pressure, and in addition is independent of the speed of rotation. If the speed is changed, mercury will automatically flow through the communicating tube to maintain balance between the centrifugal force and the air pressure. Several important performance characteristics are implied by the above:

a. A much greater force output is potentially available compared to that of comparable conventional machines such as now installed in the NOL vibration laboratory.

b. Amplitude control is simple and continuous and may be performed from a remote location if desired (this is important for tests involving significant quantities of explosive).

c. Each unit of the pair of drivers is automatically adjusted to the same force.

d. The acceleration will be nearly constant regardless of frequency if the driving force remains constant (see paragraph 7). Constant-acceleration operation makes automatic cycling feasible by eliminating the necessity for monitoring the amplitude. An equivalent to the Standard 24 hour Laboratory Transportation Vibration schedule can be run off by setting the air pressure to correspond to 2g and varying the speed uniformly from 700 to 3000 cpm by means of some automatic device, the design of which (unlike that of a device required to control both amplitude and frequency) would appear to present no great problems.

Under such an arrangement once the test is started the machine should not require the attention of the operator although the test specimen might to control damage.

General Construction

8. A precision pressure regulator with automatic bleed to permit reducing the pressure will be installed to control the driving force. The construction of the remainder of the machine is more or less conventional. Flexure-plate support of the table was considered desirable in view of the high amplitude of vibration. The flexure plates permit use of a single centrally located spring mounted on a screw-jack to maintain a definite table height under varying loads. A Speedranger unit will drive the two rotors through two gears within a gear box which maintain the rotors in their correct relative angular position and two drive shafts with universal joints and splines. Locating the gears off the table should eliminate the undesirable high frequency components introduced by gear-tooth noise in previous contra rotating-driver designs where the gear box was on the table itself. An indicating electrical tachometer completes the equipment.

9. Compressed air was selected as the control medium because of its availability from the laboratory supply. Other gases might also be used with certain advantages. Nitrogen could be obtained in cylinders at high pressure in a moisture-free condition and due to its inert nature should cause no oxidation of the mercury or driver. However, one chamber of the driver must be open to the atmosphere. The presence of a small amount of dirt or moisture should not affect the accuracy of the machine because the calibration depends only on the gas pressure. A machine could be designed to obtain a greater capacity with a given amount of mercury if a higher gas pressure were utilized. Regarding corrosion, steel is corroded only slightly by mercury and steel-mercury combinations have been used successfully in other apparatus. Mercury is not expected to remain in contact with the Oilite bearing in the shaft seal; hence, corrosion here is not anticipated. The machine is not expected to create any special health or safety hazards due to the use of mercury as this material will be entirely confined in the rotors.

Theory of Reaction Vibration

10. It is shown in textbooks on vibration, Reference (d) for example, that if a single-degree-of-freedom system consisting of a mass M supported by a spring, the deflection constant of which is K lb/in. is acted upon by a force $F \sin \omega t$ the deflection x of the mass may be expressed by the equation:

$$x = \frac{F/K}{1 - (\omega/\omega_n)^2} \sin \omega t = \frac{F/K}{1 - (f/f_n)^2} \sin \omega t \quad (1)$$

Where f is the frequency corresponding to the angular velocity ω and f_n is the natural frequency of vibration of the spring-mass combination. It is assumed that there is no damping. If x_s is the assumed static deflection of the mass under the (constant) load F .

$$x_s = \frac{F}{K} \quad \text{and}$$

$$x = x_s \frac{1}{1 - (f/f_n)^2} \sin \omega t \quad (2)$$

The single amplitude is the peak value of x :

$$\text{Ampl.} = x_s \frac{1}{1 - (f/f_n)^2} \quad (3)$$

The acceleration of the harmonic motion $x = x_0 \sin \omega t$ is $\frac{d^2x}{dt^2} = -x_0 \omega^2 \sin \omega t$.

Differentiating equation (1):

$$\frac{d^2x}{dt^2} = \frac{-F/K}{1 - (\omega/\omega_n)^2} \omega^2 \sin \omega t$$

If ω is large compared to ω_n ,

$$\frac{d^2x}{dt^2} \approx F/K \omega_n^2 \sin \omega t$$

Since $\omega_n^2 = K/M$,

$$\frac{d^2x}{dt^2} \approx F/M \sin \omega t \quad (4)$$

That is, the driving force is approximately equal to the mass times the acceleration.

Predicted Performance

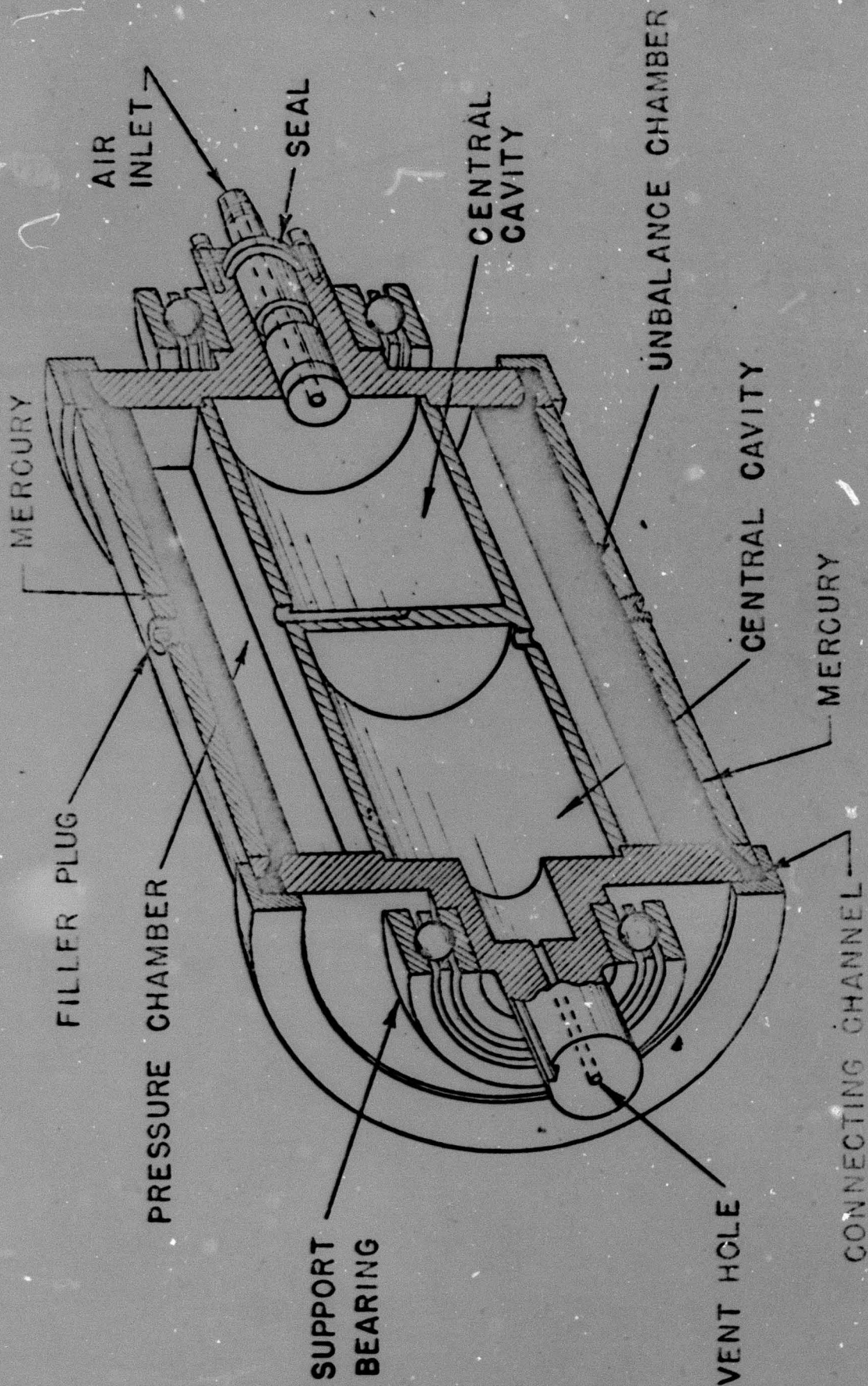
11. The driving force (f) of the two rotors corresponding to any frequency (f) and air pressure (P) may easily be computed from $F = PA$, where A is the longitudinal cross-sectional area of the chamber, and the amplitude and accelerations may then be found by means of equations (3) and (4). The natural frequency (f_n) of the table-spring flexure plate combination which enters into equations (1), (2), and (3) is given in Plate 5. The predicted performance curves have been computed for three assumed loads (1, 20 and 60 lb) and are given in Plates 6, 7 and 8 respectively. The curve of amplitude vs frequency applies to the value of the air pressure which is given; the value of the acceleration shown is approximately the average value throughout the frequency range. In Plates 6, 7 and 8, the amplitude is limited by the maximum available centrifugal force up to about 1900 cpm, above which frequency either the capacity of the bearings or the available air pressure governs the output. The machine was designed to give maximum desired performance at a pressure obtainable from the regular laboratory compressed air supply, assumed not to fall below 90 psi.

12. The maximum load is determined by the maximum available deflection of the spring. The spring would close up solid under a load of about 90 lb. Heavier test objects can be handled (at correspondingly lower accelerations) if necessary by simply replacing and single supporting spring with an appropriately stiffer one. The size of the table is 15 x 15 inches.

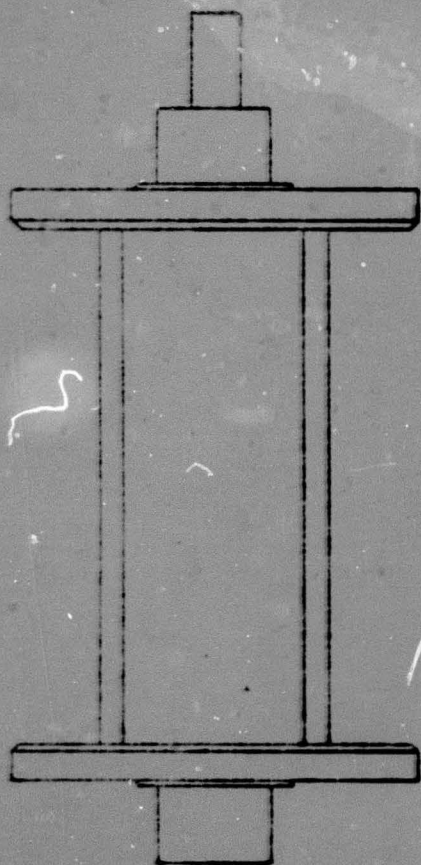
W. C. Brueggeman

J. T. Hart

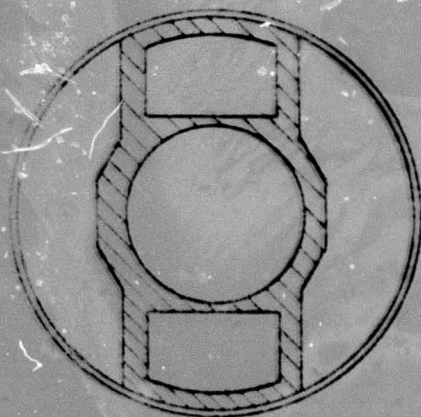
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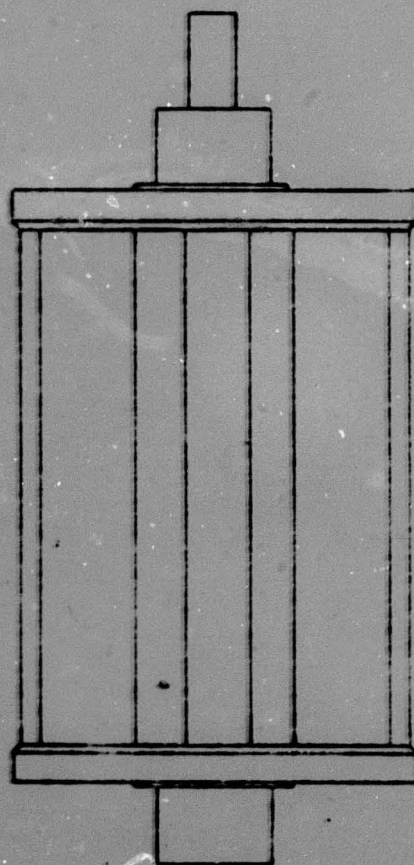
SECTIONAL VIEW OF ROTOR



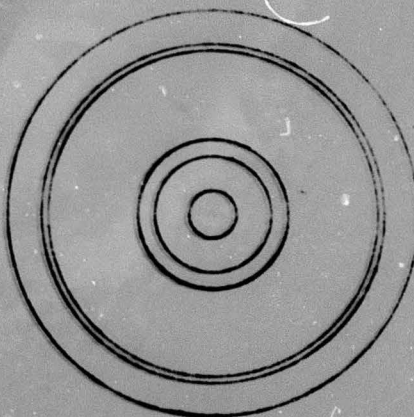
PLAN



SECTION



ELEVATION

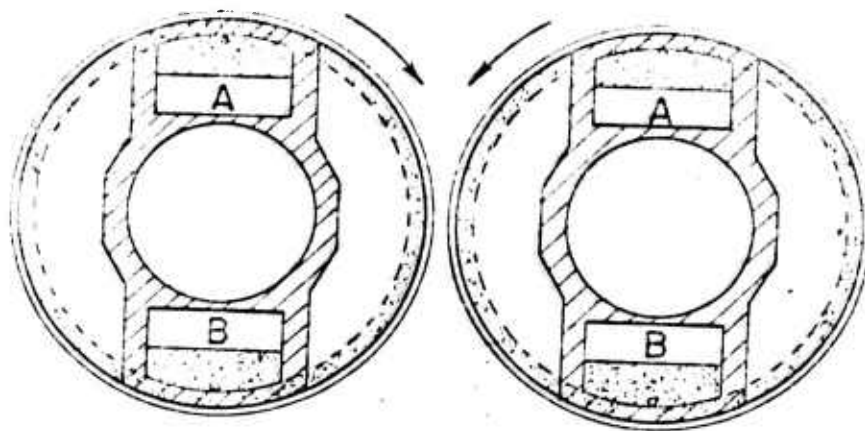


END VIEW

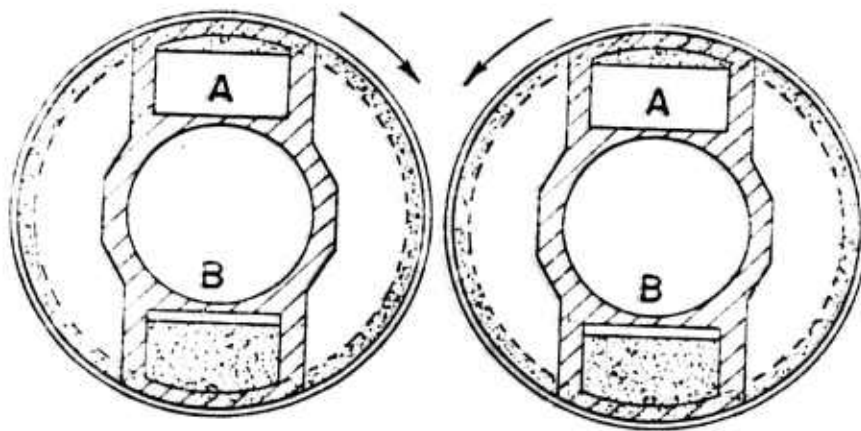
MULTITIGEE VIBRATOR ROTOR

MULTIGEE VIBRATOR

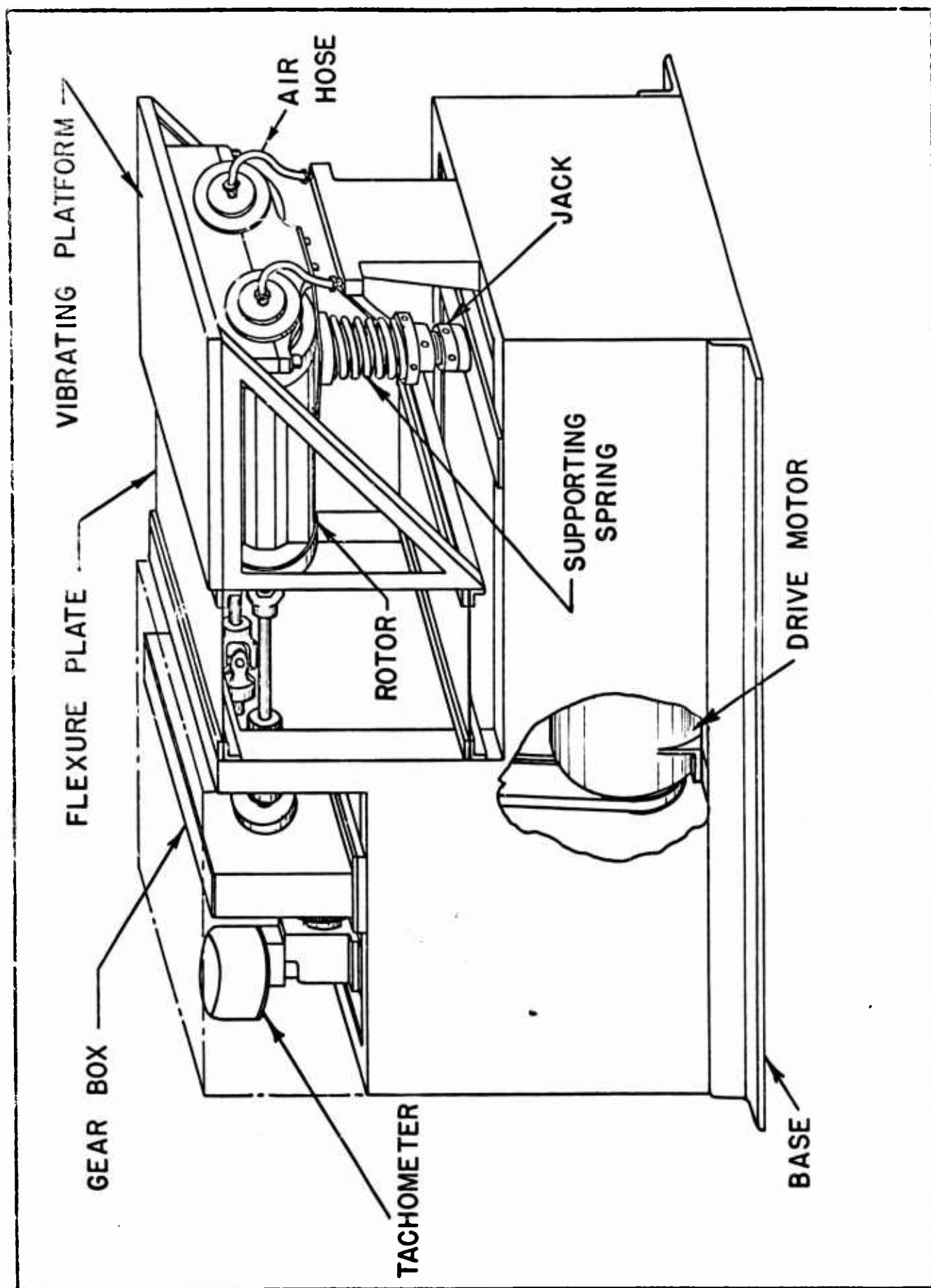
DIAGRAM SHOWING HOW ECCENTRICITY OF THE MERCURY MASS AND RESULTANT CENTRIFUGAL FORCE IS CONTROLLED BY AIR PRESSURE.



A & B UNDER ATMOSPHERIC PRESSURE -
RESULTANT CENTRIFUGAL FORCE IS ZERO.

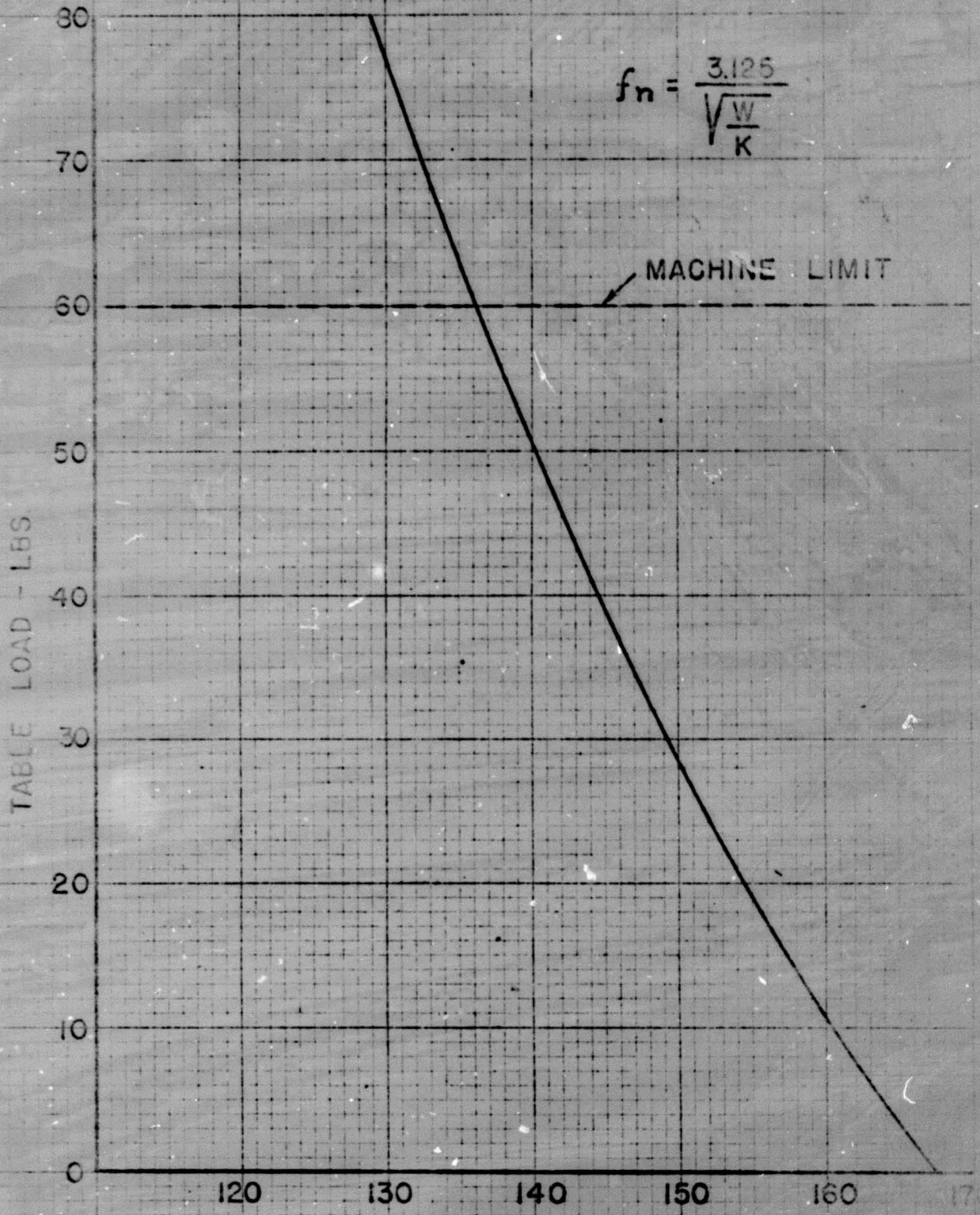


PRESSURE AT A GREATER THAN ATMOSPHERIC -
B AT ATMOSPHERIC - RESULTANT CENTRIFUGAL
FORCE IS CONTROLLED BY AIR PRESSURE.



NOLM 10248

LOAD VS. NATURAL FREQUENCY OF TABLE



NATURAL FREQUENCY OF TABLE - CPM

MULTIGEE VIBRATOR

TABLE LOAD - 1 LB.

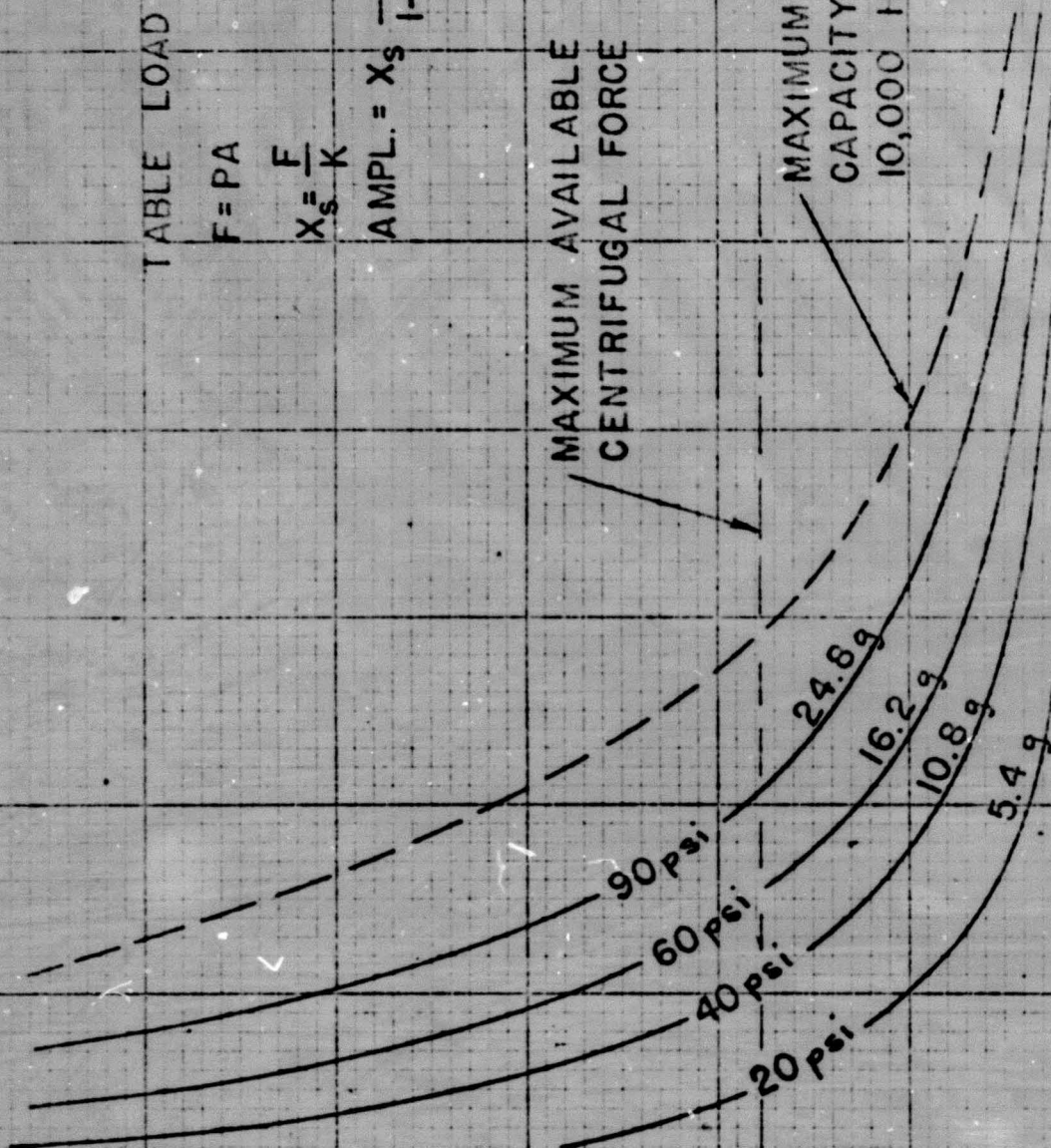
$$F = PA$$

$$X_s = \frac{F}{K}$$

$$AMPL. = X_s \frac{1}{1 - \left(\frac{f}{f_n}\right)^2}$$

MAXIMUM AVAILABLE
CENTRIFUGAL FORCE

MAXIMUM BEARING
CAPACITY - BASED ON
10,000 HOUR LIFE



FREQUENCY - CPM

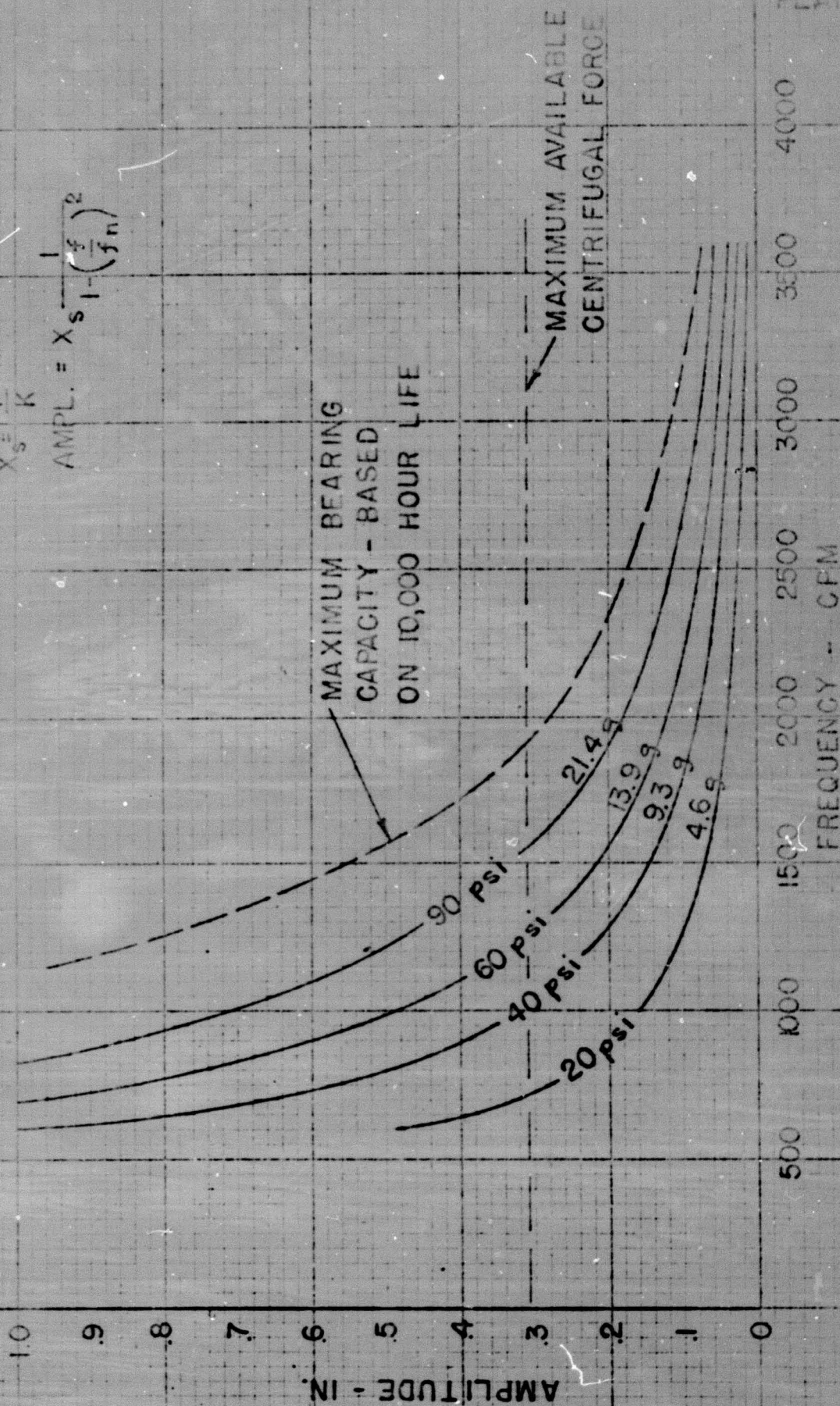
MULTIGEE VIBRATOR PREDICTED PERFORMANCE

TABLE LOAD - 20 LBS.

$F = PA$

$X_s = \frac{F}{K}$

$$AMPL. = X_s \frac{1}{1 - \left(\frac{f}{f_n}\right)^2}$$



MAXIMUM BEARING
CAPACITY - BASED
ON 10,000 HOUR LIFE

MAXIMUM AVAILABLE
CENTRIFUGAL FORCE

FREQUENCY - CFM

MULTIGEE VIBRATOR PREDICTED PERFORMANCE

MULTIGEE VIBRATOR PREDICTED PERFORMANCE

